ARMY RESEARCH LABORATORY

Rotary Engine Friction Test Rig Development Report

by Brian C. Huffman

ARL-CR-685 December 2011

prepared by

Motile Robotics Inc.
1809 Fashion Ct., Ste. 107
Joppa, MD 20185

under contract

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Army Research Laboratory
Aberdeen Proving Ground, MD 21005-5066

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14. ABSTRACT

This report describes the development work to experimentally determine the friction characteristics of the UAV Engines Limited 1100 rotary engine. Background research suggests methods to measure and reduce friction loss. The analytical equations shown are used to determine friction characteristics from experimental measurements. A computer-aided design model of an engine friction test rig was designed and fabricated to test the engine. A computer model of the rotary engine using GT-Power was created and will be validated through experimental test results. Future work includes testing, postdata analysis, and validation of a computer model.

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1. Introduction

The U.S. Army Vehicle Technology Propulsion Division (VTP) was tasked with developing an engine friction test rig to experimentally measure the friction characteristics of the UAV Engines Limited (UEL) 1100 rotary engine shown in figure 1. This is the engine currently used on the U.S. Army's Shadow 200 Unmanned Aerial Vehicle (UAV) (figure 2). A computer model of the rotary engine in development was created using GT-Power within GT-Suite, a product of Gamma Technologies. The model uses empirical parameters to determine engine friction. Experimental data collected will determine these parameters and validate the model.

Figure 1. UEL 1100 38-hp rotary engine.

Figure 2. U.S. Army Shadow 200 UAV.

Current fielded engines have a lifespan of ~250 flight hours before they are overhauled. They are overhauled three times; then, after 1000 flight hours, they are considered nonflight-worthy and are excessed. For aircraft engines, this is a very short lifespan.

The overhauls typically involve replacing the apex and side seals. To resolve this problem, fundamental research is needed to understand the friction characteristics of the rotary engine that lead to accelerated wear and tear on the seals.

Developing a friction test rig is part of an objective to develop a test facility and test bed engine for research in a power class of current interest to the Army. This will lead to developing and validating a system model for a rotary engine using GT-Power. The friction rig will also support focused research from other technical areas, including ceramic components, advanced bearing designs, etc., since an electric motor is used to spin the engine without combustion.

2. Background Research

Most rotary engine research was conducted from the 1980s to early 1990s, much of which was produced at NASA Glenn Research Center in Cleveland, OH. Relatively few technical research papers have been produced since that time. Much of the background research conducted focused on papers covering testing methods and setup for rotary engines, data analysis, and improvements in materials and components to reduce friction losses.

In Fadel et al. (*1*), motored tear-down tests are discussed, and plots of typical data and trends are shown. These will serve as reference when motored tests are done on the UEL 1100 rotary engine and test rig setup. Six oil grades were tested on a motored engine. It was interesting to note that as oil viscosity increased, engine wear decreased but fuel economy would decrease because of the increased energy waste because of the stroke being dominated by a hydrodynamic lubrication regime.

Gish et al. (*2*) performed motored and fired engine tests on four-cylinder engines with 7:1 and 12:1 compression ratios, respectively. Many charts compared the disparities between the two types of tests; an awareness of these disparities is critical for future engine testing, as is the effect of compression ratio on engine friction characteristics.

Heywood (*3*) discusses the measurement of friction mean effective pressure (FMEP) from indicated mean effective pressure (IMEP): "The gross $IMEP_g$ is obtained from $\int p\,dV$ over the compression and expansion strokes for a four stroke engine, and over the whole cycle for a two-stroke engine. This requires accurate and in-phase pressure and volume data. Accurate pressure vs. crank angle data must be obtained from each cylinder with a pressure transducer and crank angle indicator. Volume vs. crank angle values can be calculated. Both $IMEP_g$ and pumping mean effective pressure (PMEP) are obtained from the p-V data. By subtracting the brake mean effective pressure (BMEP), the combined rubbing friction mean effective pressure (RFMEP) plus auxiliary mean effective pressure (AMEP) requirements, are obtained."

Additionally, Heywood (*3*) discusses direct motoring tests: "Direct motoring of the engine, under conditions as close as possible to firing, is another method used for estimated friction losses. Engine temperatures should be maintained as close to normal operating temperatures as possible. This can be done by heating the oil flow. The power required to motor the engine includes the pumping power. In tests on spark ignition engines at part-load, the throttle setting is left unchanged. 'Motoring' tests on a progressively disassembled engine can be used to identify the contribution that each major component of the engine makes to the total friction losses."

Martyr and Plint (*4*) give an in-depth explanation of the entire engine testing process from initial design to test cell considerations to postprocessing of data. Many ideas were used to ensure standard testing practice was adhered to. Additional information on data collection and postdata analysis are discussed in Lancaster et al. (*5*). Common errors and proper techniques are identified.

Nagao et al. (*6*) presented the status of rotary engine friction as of 1987. The friction factors of the rotary engine were analyzed by the strip-down method while motoring. "Motoring" is, of course, rotating the crankshaft of the engine via an electric motor without combustion taking place. The strip-down testing resulted in FMEP contributions for the eccentric shaft, rotor, auxiliary device, gas seals, oil seals, pumping, and compression loss. A similar testing procedure will be implemented on the UEL 1100 engine.

In Ansdale and Lockley (*7*), the relationship of the stroke volume of the working chamber of a rotary engine is compared to a two- or three-cylinder reciprocating engine. This gives insight into volume vs. crank angle plots and how they are related serving as a reference. The derivation of the stroke volume, which matches Yamamoto (*8*), is shown in equation 5.

Friction loss can be decreased by optimizing the seal loading and by using lower friction materials for the seals with minimal seal wear. Friction loss can also be decreased by reduced seal area. Handschuh and Owen (*9*) did much of the analysis work on the UEL 1100 engine apex seals. Their report lays the groundwork of what to expect by experimental friction testing when comparing experimental data plots to analytical data plots.

Research by Shimizu et al. (*10*) has shown improved durability by using a two-piece apex seal design subsequently implemented into their rotary engines. These seals consisted of a fiber-reinforced and silicon-nitrided-based ceramic material. This allowed for better gas sealing than the original design. Various coating materials and coating methods have been developed to increase wear resistance and reduce friction. Though the specifics of these new designs are beyond the scope of current research, they could be customized for the UEL 1100 engine, and the friction loss changes could be measured with the friction rig. Their test data show less wear, slightly increased torque, and slightly decreased brake specific fuel consumption.

Additionally, Nagao et al. (*6*) present research to reduce wear of internal components. When micro channel porous (MCP) Cr-plating is used, the wear of the trochoid wall is reduced 50% compared with that of the conventional pinpoint porous chromium plating. To further reduce reaction and wear of apex seals under a poor supply of liquid lubricants, a fluorocarbon resin coating was applied to the MCP Cr-plated trochoid wall. This result was one-eighth the amount of wear compared to if no resin were applied.

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3. Equations

Friction equations and characteristics for rotary engines are not well understood. Thus, most friction calculations in engines are based on empirical data. Unfortunately, none of the data obtained and used to calibrate these equations were obtained from rotary engines; rather, they were for typical four- or two-stroke internal combustion engines.

The following equations will be used to determine performance properties of the rotary engine based on experimental data collected, such as static pressure, static temperature, torque, and crank angle. Afterward, FMEP can be calculated analytically to focus more on friction and other power losses.

3.1 Geometric Equations

Equations based on the rotary engine geometry need to be considered first. The working chamber volume as a function of crank angle is determined from equations in Yamamoto (*8*):

$$V_\alpha = V_{min} + \frac{3\sqrt{3}}{3} e \times R \times b \left\{ 1 - \sin\left(\frac{2}{3}\alpha + \frac{\pi}{6} \right) \right\} \ , \tag{1}$$

where

$$V_{min} = \left\{ \frac{\pi}{3} e^2 + 2e \times R \times \cos + \left(\frac{2}{9} R^2 + \ 4e^2 \right) \varphi_{max} - \frac{3\sqrt{3}}{3} e \ \times R \right\} \times b \ , \tag{2}$$

$$\varphi_{max} = \sin^{-1}\left(\frac{3}{K} \right) \ , \tag{3}$$

and

$$K = \frac{R}{e} \ . \tag{4}$$

e = eccentricity.

α = angle of shaft rotation 0°–1080°.

R = generating radius.

b = width of rotor housing.

K = trochoid constant.

φ_{max} = maximum angle of oscillation.

The eccentricity *e*, generating radius *R*, and width of rotor housing *b* are all known geometric measurements of the rotary engine. The trochoid constant *K* is an important index that determines the theoretical compression ratio and basic dimensions of a given rotary engine. The angle of rotation of the drive shaft α will rotate three times for every revolution of the rotor. This variable will be determined during experimental testing.

The stroke volume, V_H, or displacement volume, is given by the difference between the maximum and minimum values of the volume of the working chamber. It is determined in Yamamoto (*8*) as

$$V_H = 3\sqrt{3} \times e \times R \times b \ . \tag{5}$$

3.2 Performance Equations

It is common to discuss normalized work in terms of mean effective pressure (MEP), the work per cycle per unit displaced volume. Equation 6 can be found in Heywood (*3*).

$$MEP = \frac{2 \times \pi \times T \times n_r}{V_H} \ . \tag{6}$$

Power and MEP are related by (*6*)

$$P = MEP \times V_H \times \frac{N}{n_r} \ . \tag{7}$$

T is torque, *N* is the speed of rotation, n_r is the number of revolutions per cycle, which is one for two- and two for a four-stroke cycle, and *V* is the working chamber volume. The variables can vary to get brake power measured at the shaft or indicated power measured at the chamber.

IMEP can be determined as a function of crank angle degree, chamber pressure, and volume (*8*):

$$IMEP = \frac{1}{V_H} \int_{\alpha=0^\circ}^{\alpha=1080^\circ} P \times dV_\alpha \ . \tag{8}$$

Based on equation 8, the net-indicated (IMEP_n), gross-indicated (IMEP_g), and pumping IMEP (IMEP_p) are obtained by integrating over the appropriate limits.

$IMEP_n$ is obtained by integrating over the entire cycle; $IMEP_g$ is obtained by integrating over the compression and expansion phases of the cycle; and $IMEP_p$, or PMEP, can be obtained by integrating over the intake and exhaust phases of the cycle.

Once integrated using the trapezoidal rule, IMEP takes a more useful form of (*3*)

$$IMEP = \frac{1}{V_H}\sum_i \left[P(i)+P(i+1)\right]\left[V_\alpha(i+1)-V_\alpha(i)\right]. \tag{9}$$

Pumping losses are the work required to draw a mixture of air through the intake system and into the combustion chamber, and to expel the exhaust gases from the combustion chamber and out of the exhaust system. Normalized pumping losses, or PMEP, are determined by (*3*)

$$PEMP = IMEP_g - IMEP_n. \tag{10}$$

BMEP can be determined using torque data as a function of crank angle, T_α, and displacement volume (*8*)

$$BMEP = T_\alpha \frac{2 \times \pi \times 980.6}{V_H}\left(N/m^2\right). \tag{11}$$

The rotary engine has an alternator and oil pump that generate torque losses, which are measured as AMEP. This value is determined by two engine configurations, one with the alternator attached and the other with it unattached and measuring the difference reflected in the BMEP. The AMEP contribution from the oil pump cannot be distinguished from the AMEP contribution of the alternator because the oil pump cannot be removed.

Friction loss is the sum of all losses in power to the rotary engine. FMEP for the engine can then be calculated from (*3*)

$$FMEP = IMEP_g - BMEP + AMEP. \tag{12}$$

RFMEP and AMEP cannot be distinguished from each other if the alternator on/off test is not done; equation 13 would have to be used instead (*3*):

$$FMEP = BMEP - PMEP. \tag{13}$$

RFMEP can be calculated from (*3*)

$$RFMEP = FMEP - PMEP - AMEP. \tag{14}$$

Apex seals are in the plane of rotation of the engine and contribute to a large portion of rubbing friction. They are attached to the tip of the trochoid-shaped rotor and serve as a seal between the three chambers. Trochoid is like a triangle, but the sides have a curvature bulging outward from them. The exact torque loss can be determined by removing one seal and measuring the change in torque loss. This can then be correlated to the sum of the three seals.

4. Friction Rig Development

A test stand was developed to measure friction characteristics of the UEL 1100 rotary engine. This test stand included the supporting hardware and electronics required to operate the engine in the manner desired. The following instrumentation was added to obtain the required information for the friction analysis:

- A Himmelstein in-line torque meter with the sensitivity to measure torques in the range of 0–50 lbf-in. Measuring torque allows us to determine the torque load applied on the engine by various components that include the apex seals, rotor, pumping losses, and alternator.
- Type-K thermocouples to measure temperature at six locations within the engine. The temperature data will correlate oil viscosity and thermal behavior. Piezoelectric pressure transducers measure the static pressure at four locations throughout the engine: intake, compression, combustion, and exhaust.
- Three Kistler pressure transducers located within the high-temperature locations and a Kulite pressure transducer located in the low-temperature intake. See figure 3 for locations.
- An AMO optical crank angle encoder to measure the position of a pressure, temperature, or torque measurement within one-tenth of a degree over the 360° rotation in the engine. This will allow plotting of thermodynamic properties to a given crank angle within the four-stroke cycle.

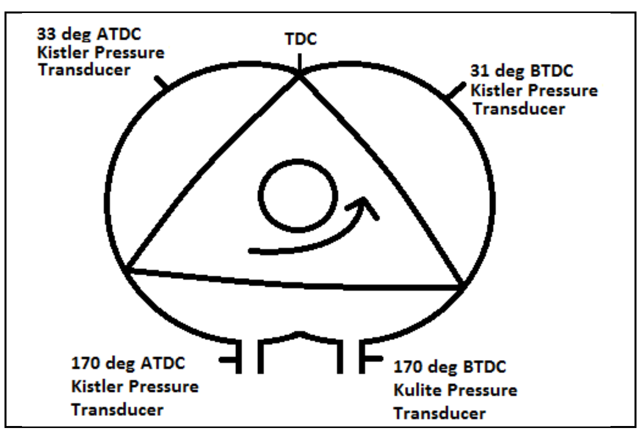

Figure 3. Pressure transducer internal mounting locations.

Note: ATDC = after top dead center, BTDC = before top dead center, and TDC = top dead center.

5. AutoCAD Model Development

A model of the rotary engine friction test rig was developed to determine the optimal placement of instrumentation while maintaining a compact footprint (figure 4). The hardware was arranged to measure all desired properties. Several iterations of refinement were done to address technical issues and space confines. The model was designed to have two levels instead of one; this reduced the length of the rig by almost half and reduced dead space underneath the rig since one design constraint was to place the engine at least 3 ft above the floor to ensure accessibility. A transparent blast shield was added for protective requirements and to allow visibility during rig operation. Since the 20-hp electric motor is one of the largest components of the rig, its actual size was modeled. This allowed us to determine the minimum height of the upper plate surface so that the electric motor would fit underneath.

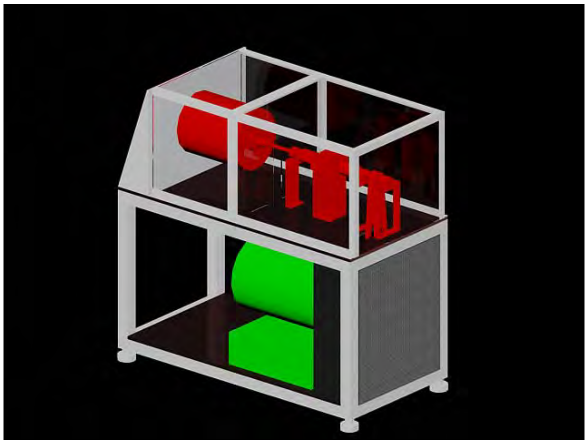

Figure 4. Engine friction test rig AutoCAD model.

Not everything was modeled in detail, so that the technician could make expert decisions based on significant experience on bolt sizing, instrument supports, and other small adjustments during construction. A separate model of a shaft-to-engine adapter was created so that it could be fabricated by the technician.

6. Physical Rig Development

After the optimal configuration was designed, the physical rig could be fabricated (figure 5). The metal frame with an upper and lower surface plates model was sent to a machine shop to be fabricated. A Vehicle Technology Directorate (VTD) technician then mounted all of the components to the frame at the desired locations as shown on the AutoCAD model.

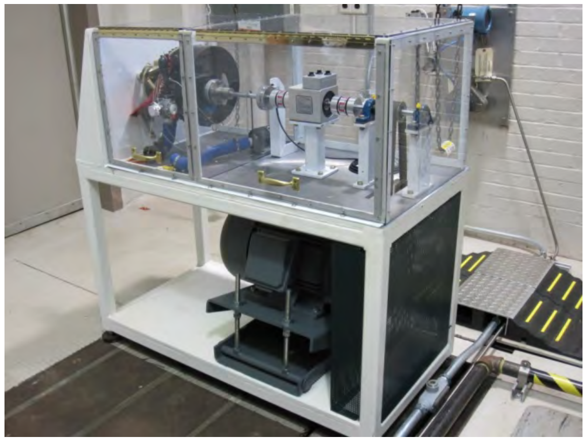

Figure 5. Engine friction test rig.

An electric motor is connected by belt and pulley to the shaft extended from the engine crank. This shaft rotates the engine's rotor through the four-stroke cycle. As the shaft rotates, an in-line torque meter measures the resistances within the engine opposing the motion of the shaft. This resistance torque is the friction loss.

A variable frequency drive controls the electric motor speed. Two pillow block bearings are used to support the shaft attached to the engine. A custom flange connects the shaft to the engine crankshaft. A protective shrouding encloses the belt and two-pulley system to prevent foreign objects from being caught within. A transparent blast shield made of a polycarbonate called Makrolon, capable of containing an 8000-rpm triburst, encloses the rotating machinery. The thickness required was verified by triburst analysis.

7. GT-Power Model Development

GT-Power is an engine modeling software used widely within the automotive manufacturing community. This development tool models internal combustion engines that are spark ignition or diesel. The four- or two-stroke engines run on a variety of fuels. Engine performance, such as power, fuel consumption, exhaust emissions, etc., can be determined. However, for accurate results, a GT-Power model needs to be calibrated using experimental data. This data is not yet available at VTD.

The UEL 1100 rotary engine was initially modeled in GT-Power as a three-cylinder, two-stroke engine (figure 6) by Handschuh and Owen (*9*). The software doesn't specifically support rotary engines but was determined to be acceptable. GT-Power is capable of simulating the engine operating conditions to a comparable extent. However, since experimental data was lacking, the default model parameters were used. As a result, the model engine operation did not match the manufacturer's stated performance.

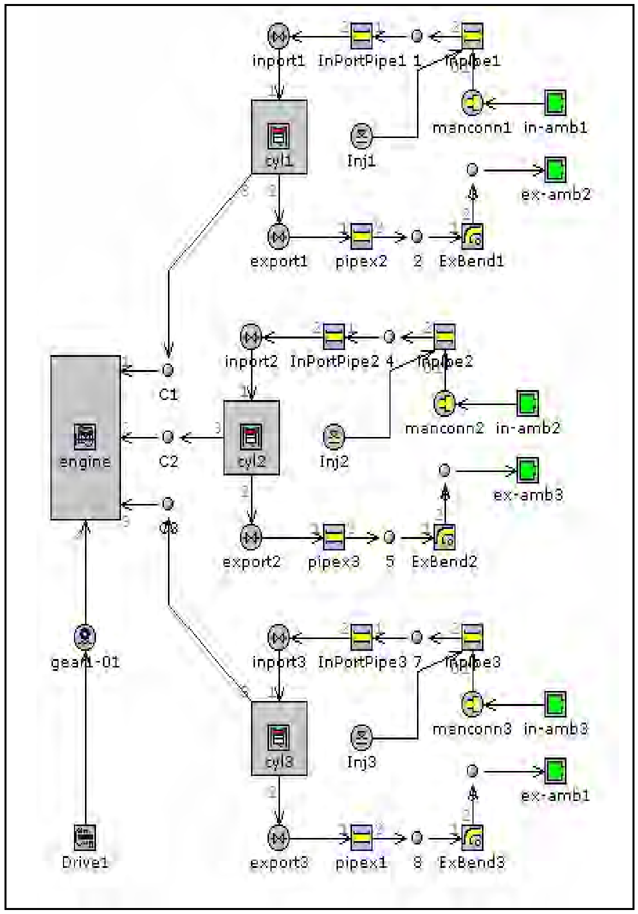

Figure 6. GT-power model of the UEL 1100 rotary engine.

Several empirical parameters must be evaluated for the model. The friction is unknown but can be set to reflect the rated 38-hp (28.3-kW) output; however, experimental data is needed to determine the actual friction. Torque measurement data can be used to determine the necessary values. Heat release and heat transfer rates are also unknown. Currently, the model uses a standard heat release package based on the properties of mogas. The data from measuring temperature can aid in resolving this issue.

Karl Owen from ref. 9 also created a modified GT-Power model that includes a turbocharger. Once the original GT-Suite model is validated, the turbocharger model will be more accurate. This validation will prepare for turbocharger and fuel-injector testing, which will lead to further development and calibration of the model. Further details are beyond the scope of this report.

8. Future Work

All performance characteristics will require experimental data from the developed rotary engine friction test rig and data acquired from a similar VTP project where the same rotary engine is operated on a dynamometer. Data acquired will include pressure, torque, temperature, and crank angle. Once all experimental data are acquired, plots of performance and friction characteristics will be generated. Based on the knowledge gained from that and additional data analysis, we can determine exactly how to implement data into the GT-Power model for validation.

All of the instrumentation needs to be connected to their respective amplifiers, controlling computer, and data acquisition device. Electrical connections between the power source, electrical motor, and variable frequency drive need to be done as well. Once all connections are made, we can verify the data acquisition setup and conduct trial runs. Then a series of tests will be conducted to gather data.

A baseline test over the entire operating range of the engine is one of several that should be conducted. From there, a test to seal the intake and exhaust ports will be conducted. Breakdown testing will consist of removing the alternator and running, removing an apex seal and running, and removing a side seal and running. Potential tests would be to replace the steel ball bearings with ceramic ones that may be lighter and/or have a lower coefficient of friction rating. Last, a potential test would use alternative lubricating oils with various viscosities.

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List of Symbols, Abbreviations, and Acronyms

α	angle of shaft rotation
AMEP	auxiliary mean effective pressure
ATDC	after top dead center
b	width of rotor housing
BMEP	brake mean effective pressure
BTDC	before top dead center
e	eccentricity
FMEP	friction mean effective pressure
GT	Gamma Technologies
IMEP	indicted mean effective pressure
$\mathrm{IMEP_g}$	gross indicated mean effective pressure
$\mathrm{IMEP_n}$	net indicated mean effective pressure
$\mathrm{IMEP_p}$	pumping indicated mean effective pressure
K	trochoid constant
lbf-in	pound-force inch
MCP	micro channel porous
MEP	mean effective pressure
n_r	number of revolutions per cycle
N	Newton, rotation speed
NASA	National Aeronautics and Space Administration
φ_{max}	maximum angle of oscillation
P	pressure
PMEP	pumping mean effective pressure
R	generating radius

RFMEP	rubbing friction mean effective pressure
T_α	torque per crank angle
TDC	top dead center
UAV	Unmanned Aerial Vehicle
UEL	UAV Engines Limited
V_α	volume per crank angle
V_H	displacement volume
VTD	Vehicle Technology Directorate
VTP	Vehicle Technology Propulsion Division
W_c	work per cycle

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